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VAN. H. MANNING, DIRECTOR

ECONOMIC OPERATION OF STEAM TURBO-ELECTRIC STATIONS

BY

C. T. HIRSHFELD AND C. L. KARR

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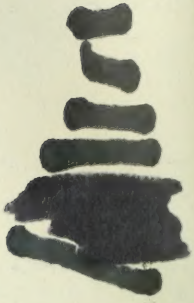

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ECONOMIC OPERATION OF TURBO-ELECTRIC STATIONS.

By C. T. Hirshfeld and C. L. Karr.

INTRODUCTION.

Under war-time conditions, with the demand for fuel taxing the productive capacity of the mines and exceeding the facilities for quick distribution, there is especial need of giving the utmost attention to all methods whereby loss or waste of fuel may be minimized or prevented and all heating or power plants worked at highest efficiency.

The large central power or heating plants, such as are now in service in many cities, offer special opportunities for practical application of fuel-saving methods, because the increase of economy can be had by making changes that are practicable within a reasonable time and do not call for the employment of higher engineering supervision than is already available; whereas the losses at small plants, although vastly greater in the aggregate, require the education, training, and supervision of boiler-room employees and, at many plants, the replacement or extensive modification of present equipment.

In a series of publications[a] the Bureau of Mines has discussed the various problems involved in obtaining higher furnace efficiency at the smaller heating and power plants, especially those with hand-fired boilers. The present paper discusses the economies to be effected in the use of fuel through changes in operating methods at large power plants having turbogenerators.

[a] Flagg, S. B., Cook, G. C., and Woodman, F. E., Experiments with furnaces for a hand-fired return tubular boiler: Tech. Paper 34, Bureau of Mines, 1916, 32 pp.

Clement, J. K., Frazer, J. C. W., and Augustine, C. E., Factors governing the combustion of coal in boiler furnaces, a preliminary report: Tech. Paper 63, Bureau of Mines, 1914, 46 pp.

Kreisinger, Henry, Hand firing soft coal under power-plant boilers: Tech. Paper 80, Bureau of Mines, 1915, 83 pp.

Kreisinger, Henry, Ovitz, F. K., and Augustine, C. E., Combustion in the fuel bed of hand-fired furnaces: Tech. Paper 137, Bureau of Mines, 1916, 76 pp.

Flagg, S. B., Firing bituminous coals in large house-heating boilers: Tech. Paper 180, Bureau of Mines, 1917, 22 pp.

Kreisinger, Henry, Ovitz, F. K., and Augustine, C. E., Combustion and furnace design: Bull. 135, Bureau of Mines, 1917, 144 pp.

FACTORS DETERMINING FUEL ECONOMY.

The maximum economy attainable in the use of fuel in any given turbogenerator station is determined by the characteristics of the apparatus installed and by the way in which that apparatus is assembled and connected—that is, by the design of the station.

The actual working economy is determined by the fuel used, by the temperature of available circulating water, by the character of the load to be carried, and by the operating methods adopted.

Under normal conditions that fuel should be used which gives the minimum figure for the sum of the following items:

1. Cost of coal delivered to storage.
2. Cost of preparing coal for use, as by crushing or screening.
3. Cost of transferring coal from storage to bunkers.
4. Cost of transferring coal from bunkers to furnaces.
5. Cost of removing refuse.
6. Cost of rehandling and other labor caused by spontaneous combustion.
7. Maintenance of coal-handling, preparing, and burning equipment.
8. Capital charges on equipment required to generate the required amount of steam.

In the present state of knowledge there is no method of determining, by reasoning or by analysis, the coal that will give this minimum. In general, each variety available in a given district must be given a trial thorough enough to enable the operating engineer to obtain actual cost figures. As a rule a medium-price coal gives the best results. Occasionally the coal that is most expensive on a ton basis really gives the minimum total of all costs concerned.

The temperature of the circulating water is, in general, entirely dependent on the characteristics of the supply and on the effect of seasonal changes and is thus beyond the control of the operating engineer.

The character of the load to be carried is also, in general, beyond the control of the operating engineer. He must accept that load which the industry or community that he serves gives him at any particular instant. His job is to carry that load throughout all of its variations in such a manner as to attain as nearly as he can the minimum production cost possible with the equipment available, but he must always carry this load in such a way as to obtain the degree of continuity of service demanded or desired.

Operating methods of this sort require the closest study of all of the items entering into the cost of net or available power—that is, power leaving the plant and available for external use. Labor, supplies, and maintenance in turbine room and boiler room are just as

important as is fuel cost. No detailed operating method can be laid down as applicable to each plant, because apparently insignificant local pecularities may in reality greatly modify the most economic operating method.

This paper is intended to consider only the effect of operating methods upon the fuel consumption, and the parts following are devoted entirely to that phase of the problem. However, in reading them, one must bear in mind constantly the more general considerations already enumerated. Methods giving the smallest fuel charges are not necessarily the most economical methods: true economy comes only from a realization of the complete problem and the use of an operating method which represents the best possible compromise between the many conflicting interests outlined above.

In a single plant with a given load curve, the fuel economy is dependent upon three distinctly different things. These are:

1. The method used in distributing the load between the main units available;

2. The methods used in the boiler room for producing the steam required; and

3. The method used for obtaining auxiliary power, or house service.

These three branches of the problem will be considered separately in what follows.

DISTRIBUTION OF LOAD BETWEEN MAIN UNITS.

Most of the steam-driven turbogenerators built in this country have steam characteristics of the general type shown in figure 1. The steam required, per kilowatt-hour delivered, decreases with increasing load until some best point is reached, after which increasing load causes an increase in the steam used per unit of power delivered. Or, the total steam consumption increases approximately according to a straight-line law until at some point the increase occurs at a more rapid rate, causing the graph of total steam consumption to curve upward as shown in figure 1. Some turbines built recently possess steam characteristics giving more complicated graphs when plotted,

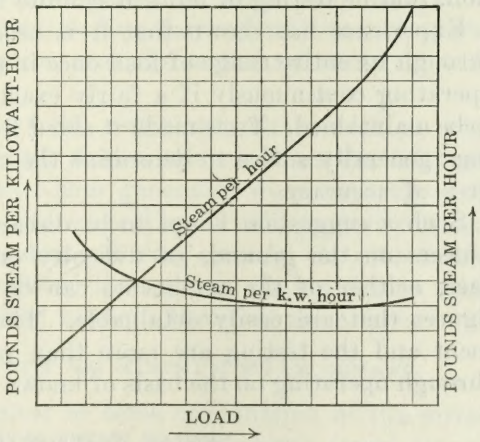

FIG. 1.—Steam curves for turbogenerator.

but attention will be confined to the type shown in the figure because of the greater simplicity and because the same sort of reasoning can be used to evolve proper operating methods for units with more complicated characteristics.

WATER-RATE CURVES.

It is well known that steam pressure, superheat, and vacuum or back pressure all have marked effects upon the steam consumption, or water rate, of steam turbines. The water rate is decreased by increased pressure, by increased superheat, and by improved vacuum. Pressure and superheat may be regarded as fixed in any given plant, but the vacuum attained is of necessity quite variable, when considered over long periods of time.

It is therefore possible to obtain a whole family of water-rate curves for a given machine in a given station, each member of the family corresponding to a different vacuum, or back pressure.

Moreover, the water rate of a given unit under given conditions of pressure, superheat, and vacuum also varies from time to time. The variations are apparently due to such things as deposits on blading, wear of blading, variation of clearances, settlement in bearings, and other uncontrollable factors. The exact shape and position of the family of curves mentioned is therefore constantly shifting.

Any logical method of operating turbogenerators in such a way as to cause them to make minimum steam demands for given loads must obviously depend upon a fairly complete knowledge of the shape and location of their water-rate curves. These can be obtained only by water-rate tests, and it therefore follows that, for best operation, routine testing of units at definite intervals is necessary.

Experience has shown that it is necessary to test each machine through its entire range of load once in every eight weeks at stations operating continuously if a fairly exact knowledge of conditions is to be maintained. Tests made at about five or six equally spaced loadings generally suffice to determine the curves with the required degree of accuracy.

Such a suggestion is apt to be discarded by many operating engineers on the grounds of difficulty or expense. As a matter of fact, neither of the arguments can be maintained in the face of figures that are easily obtainable. The cost of the testing equipment and the testing are more than paid by the savings effected through operating on the basis of known water rates.

NEED OF WATER-RATE TESTS.

Some of the recently built stations using surface condensers have incorporated the weighing tanks, scales, and piping necessary to per-

mit the crew to make weighed water-rate tests at a minute's notice. The work consists merely in opening and closing a few valves and taking the necessary readings. Tests of this sort are very accurate if made at fairly steady loads over long enough periods of time, and if condenser leakage and steam leakage are guarded against. Condenser leakage is detected easily by well-known chemical methods if the circulating water is salt, and is just as easily and more expeditiously found if the circulating water is fresh by determining the electrical conductivity of the condensate.

At plants having surface condensers that were constructed without provision for tests of water rate by weighing, it is often possible to install the necessary equipment at moderate expense. Particularly, when all hot-well pumps discharge into one header, it is usually a comparatively cheap and simple matter to install a parallel header with valves so arranged that any hot-well pump can discharge into weighing tanks while the discharge of the other pumps travels the customary route.

In surface condenser plants in which weighing apparatus can not be installed, or in which water-jet air pumps are used, and in plants fitted with jet condensers, recourse must be had to steam meters of some sort. The results obtained in this way are not as accurate or trustworthy as those obtained by weighing, but they can be made accurate enough to yield results that will serve as a basis for operating methods.

It is therefore justifiable to assume that operating engineers who care to obtain the best possible results can make provision for such periodic testing as is necessary to give them a fairly complete knowledge of the water-rate curves of the units under their charge. If this is so, it is certainly rational to base further treatment of operating methods on the assumption that the necessary water rates are known.

The characteristic shape of the water-rate curve, with its well-defined minimum, shows plainly that a unit consumes the minimum quantity of steam per kilowatt-hour generated when running at the load giving that minimum. Moreover, the curve is generally of such a shape that the water rate does not vary greatly for considerable variations of load either side of that giving the minimum value. The operating method outlined below is based on these facts.

COMBINING STATION UNITS FOR MINIMUM STEAM DEMAND.

It is obvious that there must be some combination of the several machines generally available that will cause minimum steam demand for any given load. It is often casually assumed that this distribution is that which causes all the machines in operation to carry loads as nearly as possible approximating those that correspond with the

minimum points of their respective load curves. This assumption is only correct when all machines have curves of the same shape and height; that is, when all have practically the same water-rate curve.

For such conditions, the minimum steam demand is obtained by operating such a number of machines as will be required to carry the existing load when each of those operating is carrying a load near its best point. Thus assume a station containing two 10,000-kilowatt units and one 15,000-kilowatt unit, all having exactly the same water rate curve. Assume that for the conditions existing the two smaller machines have their best point at 7,000 kilowatts and that the larger has its best point at 11,000 kilowatts.

If a load of 10,000 kilowatts is to be carried, two of the small machines might be operated at 5,000-kilowatt loads; or the large machines might be operated alone; or the larger machine and one of the smaller might be operated in parallel with any desired distribution of load between them. With water-rate curves of usual shape it would be found that the single large unit operating at 10,000 kilowatts would create a smaller steam demand than would the two smaller machines operating at 5,000 kilowatts each.

Also, either of these combinations should make smaller steam demands than any possible combination of one large and one small machine under the assumed conditions. The choice would then probably be between the first two possibilities.

If continuity of service was not of great importance the operation of the one large unit would be the correct solution. With such an arrangement, however, an accident that disabled the one unit operating would completely interupt the service. Therefore, when the operator is limited to obtaining the minimum steam consumption consistent with reasonably certain continuity of service he must choose to operate the two smaller machines despite the slightly greater steam consumption. An accident to one would probably cause no interruption because the other could pick up all of the load dropped by the disabled machine and still not exceed its own load-carrying ability.

If a load of 15,000 kilowatts is to be carried it is perfectly obvious that the minimum steam demand will result from the use of the two smaller machines, each carrying 7,500 kilowatts. This arrangement is not consistent with reasonable assurance of continuity of service because the surplus capacity of one machine is not enough to absorb the load carried by the other in case of failure. As a matter of fact, continuity of service at this load could not be assured by any combination of two machines, and if assured continuity were considered of great importance three machines would have to share the load. The best distribution under such condition could be determined easily by assuming different values for the load carried by the largest

machine, dividing the remainders equally between the other two and then determining the steam demand with each combination.

Under the assumed conditions of like water-rate curves for all units the solution of the problem of obtaining assured continuity of service and low steam demand becomes simpler as the number of units available is increased and as the capacities of the units approach one common value.

In practice it never happens that all units available have the same characteristic curves. Even if they are all of one design and of the same size, it will be found that their water-rate curves differ materially. The problem actually met is therefore more complicated than that outlined above, but the principles involved in its solution are about the same.

METHOD OF DETERMINING LOAD DISTRIBUTION.

It is possible to determine mathematically the distribution of load between units so as to yield the minimum water rate for any given station load, but this method has not yet been reduced to a form easily understood and applied. An alternative cut and try method is therefore given here. It is less exact and is time consuming, but it is capable of giving excellent results if applied carefully.

For the purpose of illustration, consider a station containing six units designated as A, B, C, D, E, and F, with water-rate curves for given conditions approximately as shown in figure 2. Three of these machines might be called 10,000-kilowatt units, two 14,000-kilowatt units, and the largest a 20,000-kilowatt unit, but it is important to note that what they are rated or called has nothing to do with the problem in hand. The curves show that each machine is capable of carrying a load up to some particular value and that it uses a certain different number of pounds of steam per kilowatt-hour at each different load up to that maximum value.

For convenience in reading, the horizontal lines *a*, *b*, *c*, etc., have been drawn. The line *a* is tangent to the lowest point of curve E, that showing the water rate of the unit second in economy. The line *b* has been drawn tangent to the lowest point of curve C, that of the unit ranking next, and so on.

It is not probable that a station of the size indicated would carry loads as small as 2,000 kilowatts at any time, but the individual curves have been plotted down to that value and the analysis will be started at that point for the sake of completeness. Inspection of the curves shows that machine C has the lowest water rate at 2,000 kilowatts and therefore is the machine which would be used for carrying a load of that magnitude.

If a straightedge be placed across the figure horizontally at a height representing 14.5 pounds of steam per hour and be gradually

moved downward while kept parallel to the horizontal axis it will cut all curves at equal water rates. Comparison of loads at different equal water rates on the several curves shows that machine C can carry any load from 2,000 kilowatts up to about 9,000 kilowatts more economically than any other machine.

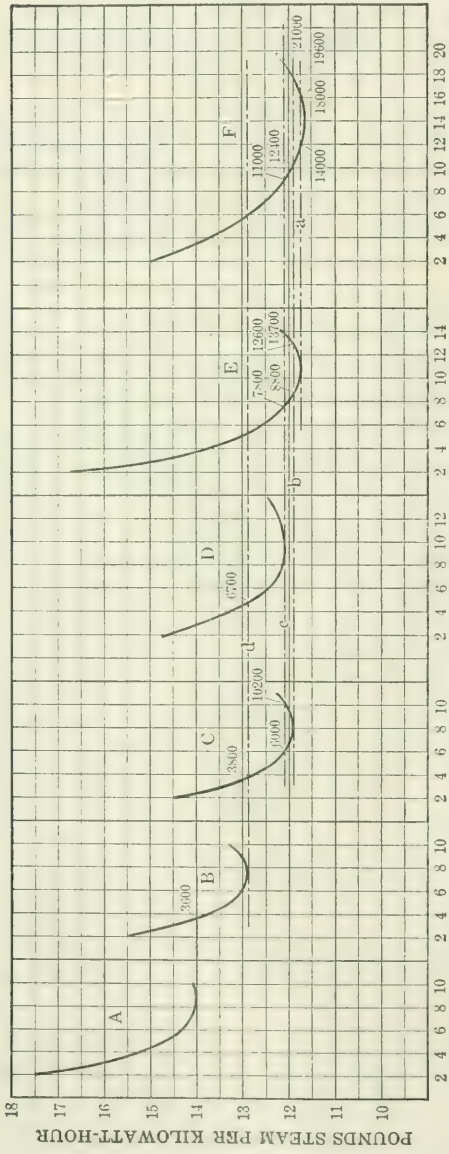

Fig. 2.—Water-rate curves for six turbogenerator station units, showing relations of load and steam consumption.

Beyond this point machine E should be given the load. It can carry loads from 9,000 to 12,500 kilowatts more economically than any other machine.

Beyond that point the load should be given to F, and it is obvious from the figure that this machine is fitted to carry from 12,500 up to 18,000 kilowatts more economically than any other.

Next, it is necessary to decide whether a load above 18,000 kilowatts can be carried more economically on F alone or on some combination of machines.

Actual numerical trial will show that the greatest economy will be obtained by carrying load on F up to some point above 18,000, but that somewhere between 18,000 and 19,000 kilowatts the load should be shifted to C and E, giving about 8,000 to C and about 11,000 to E. Such a combination would yield a water rate of about 11.8 at 19,000, as against about 11.85 for the same load on F alone. The point of change may be assumed at about 18,500 kilowatts.

It is next necessary to discover how far to carry the load on the combination C and E. Inspection of the curves indicates that 8,000 kilowatts on C and 12,600 on E will give a load of 20,600 kilowatts with a water rate corresponding to the height of the line *b*. Inspection also shows that 8,800 on E and 12,400 on F will give a total load of 21,200 with the same water rate.

Obviously the load would be carried on C and E from a total of 18,500 to about 21,000 after which it would be carried on E and F to a total of 12,600+19,600=33,200 kilowatts. The best distribution between E and F at each different load between 21,000 and 33,200 must be found by cutting and trying, but it is obvious from the curves that at loads between 11,000+14,000=25,000 and 11,000+18,000=29,000 the load will be held constant at 11,000 on E and the remainder will be taken on F.

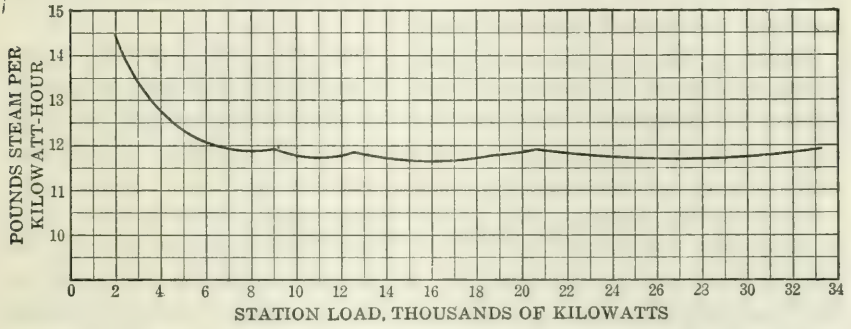

FIG. 3.—Water-rate curve for different station loads.

Beyond a load of 33,200 kilowatts the possibility of using machine C must be considered because the water rate when carrying that load on E and F in parallel is the same as the best water rate on C. As a matter of fact the load would have to be increased beyond 33,200 before C is utilized because this unit can not be utilized with the water rate in question at a load less or greater than about 8,000 kilowatts.

This sort of analysis can be carried through in the way outlined until a loading schedule has been developed for all loads between 2,000 kilowatts and the maximum of which the station is capable. It is laborious and slightly confusing when more than three machines are being considered, but the fuel savings that can be effected by operating on schedules developed in this way are so great as to amply justify much more labor than can possibly be involved in utilizing this method.

The water rates for different station loads that result in the case assumed above have been plotted in figure 3 up to the point to which the analysis was carried. It should be noted that the curve gives

only the water rates resulting from the operation of the main units and that the real station water rates will be further modified by steam used for heating, warming, and starting units, and for operating auxiliaries.

It is also interesting to note that the curve is far from smooth but that it shows a gradual downward trend to a load of 16,000 kilowatts. Beyond that point it must show a gradual rise because of the necessity of utilizing machines at less economical points or less economical machines in order to obtain the greater capacity.

In the analysis as carried out attention was given to one point only, namely that of obtaining minimum water rates at each load. In practice other considerations must be borne in mind. Thus, operating requirements do not permit the frequent starting and stopping of machines to meet a violently swinging load, nor would such frequent starting and stopping give the results desired because of the necessary incidental waste of heat.

Again, in the analysis, one machine was assumed to operate alone at the smaller loads and to be used if necessary well up toward its maximum capacity. It is possible that some stations might be willing to operate in this way, but if assured continuity of service was desired such operation would not be permissible.

Further, the arrangement of auxiliaries may often be such as to change the conditions decidedly. That arrangement giving the greatest economy, in so far as the main units are concerned, may be far from the best when the thermal cost of auxiliary power is taken into account.

Such considerations as well as others of a more or less local nature must always be given full weight. It is generally most convenient to prepare an operating schedule as suggested above and then to modify this, at various points, as necessary to meet other conditions.

It is also essential to bear in mind the fact that in normal cases schedules of this sort are good for about two months only. After that they must be revised, on the basis of experimentally determined curves, to meet the new individual water rates produced by changes in the temperature of the circulating water and by changes in the conditions of the turbines themselves.

The most advantageous distribution of load between stations can be determined in the same way as has been suggested for the distribution of load between the machines of one station. When two or more stations are contiguous or are so operated that they may be treated like parts of the same station, all of the units may be considered as separate entities and the analysis performed just as was done above.

When, however, the stations are operated independently and simply share the load as dictated by a system operator or other corresponding individual it is more convenient to proceed in a different

way. The water-rate curve for each station as a whole is drawn in about the same way as shown in figure 3 (p. 13), and these total curves are then handled as were the curves of individual machines in the preceding analysis. This method does not give the lowest water rate theoretically obtainable with perfect freedom in making all desirable combinations of all of the individual machines in all of the stations, but in practical operation it often yields more satisfactory results.

The same sort of analysis is obviously applicable to the case of transfer of load between independent companies with connected net works. Here the problem is still further complicated, in general, by the difficulty incident to obtaining complete information with regard to the characteristics of both stations and the further difficulties experienced in obtaining the sort of cooperation between independent organizations which is necessary to utilize to the full the results of such analyses.

Moreover, the electrical characteristics of the transmission network must be taken into account when distributing load between two or more power plants or systems. Losses of various sorts in the network often make it necessary to apportion loadings differently than would result from a consideration of the thermal characteristics of the several plants concerned.

BOILER-ROOM OPERATION.

The boiler-room operation, like turbine-room operation, should be based upon the characteristic curves of the apparatus available. Here the characteristic curve of importance is one showing the values of the combined efficiency of the boiler unit at all different loads within its operating range. By combined efficiency is meant the value obtained by dividing the heat supplied the material passed through the boiling vessel in a given time by the total heat in the fuel passed through the furnace in the same time.

This curve has a characteristic contour; it is always domed like the curves shown in figure 4. The exact shape of the curve, its height, and its extension to left and right vary with the proportions and type of apparatus and can be determined only by careful experiment in each case.

The word careful is used advisedly. In the first place, accurate water measurements and accurate coal measurements are necessary. Second, careful sampling and analysis of coal are essential. And, third, the tests must be conducted with full appreciation of the differences between operating conditions and test conditions as ordinarily understood.

If the tests are to be of maximum use in the operation of the boiler room they should not be run under conditions that would be expected to give results better than those attainable with the exercise

of the average amount of skill and patience of the operating crew in the daily routine of plant life. Tests made under better conditions are interesting as showing results toward which to aim, but they have not the same utilitarian value as tests made under conditions that as nearly as possible represent the best that can be reasonably expected as maintained performance.

For example, the water-washed surfaces of the boiler should be in average condition; that is, covered with about the quantity of scale and mud that would be expected about halfway between cleanings. Again, the gas-washed surfaces should be cleaned to the same extent and at the same time intervals as in daily routine operation. Further, the fireman or operator should not be the best, nor the poorest available; he should represent about the average operator in the plant.

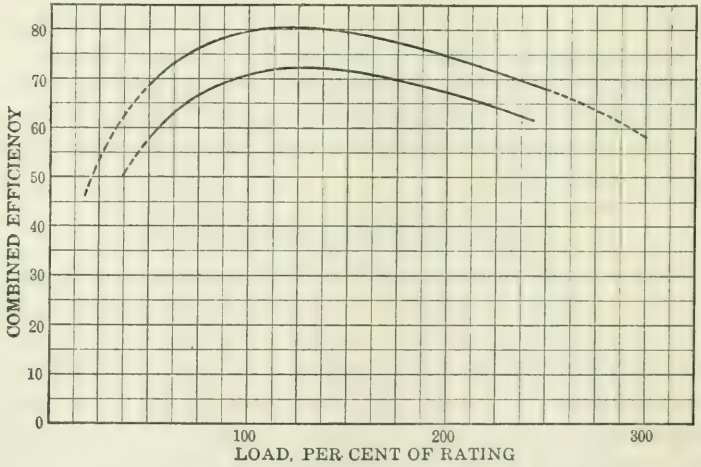

FIG. 4.—Curves showing combined efficiency of boiler unit at different loads.

Tests run under such conditions may not give results approaching advertised records but they have the virtue of telling very nearly the average truth with regard to possible operating results.

The upper curve, given in figure 4, represents good practice with large boilers operated without economizers and with underfeed stokers a little larger than the average for a given amount of boiler surface. The values shown could be obtained under operating conditions, with skillful operators, supplied with complete sets of instruments for guiding them in the making of adjustments. It is probable, however, that few operating crews could be kept so very efficient as to maintain conditions giving a curve of this height. Values 2 to 3 per cent lower would in most cases represent extremely good practice. The great majority of operating crews would probably obtain curves from 5 to 10 per cent below that shown.

Whatever the actual height, the general shape of the curve for stoker-fired boilers should not be very different from that illustrated. This shape is of great importance. It will be noted that the efficiency falls off much less rapidly at loads above that giving the greatest efficiency than it does at loads below that point. Moreover, the dome is relatively flat, and fairly large variations of load near the best point can be obtained without very great variations in thermal efficiency. In the case illustrated, an efficiency of 75 per cent or better is obtainable between about 70 per cent and 190 per cent of rating, and 78 per cent or better is obtainable between about 85 and 160 per cent of rating.

The most economical thermal results are obviously obtainable when such a number of boilers is operated as will require those operating to carry loads near that giving the best point on the curve. It is equally evident that when swinging loads are considered it is best to have the average load on a boiler fall above rather than below the best point because of the less rapid drop of the right hand side of the curve.

Economic considerations, other than those involving the cost of fuel, also dictate average operation at a point above that giving the highest thermal efficiency. As regards apparatus of the type assumed in obtaining the curve in figure 4 an average load near 140 per cent of rating would probably give about the best compromise under normal central station conditions, unless fuel was far above average prices.

Two further modifying conditions must also be noted in this connection. First, it may be necessary to operate in such a way that peak loads of large size can be picked up quickly on boilers already in operation. This may dictate operation at a point lower on the curve than would otherwise be chosen. Second, the character of the station load may be such that very light loads must be carried during many hours out of the 24. This may dictate average operation at a higher point on the curve than would otherwise be chosen, in order that all of the boilers operating can follow the load downward for a considerable distance before banking of any boilers is required.

Banking of boilers should be carefully studied in every case. A certain definite quantity of coal is consumed during every banking period and this quantity can be determined by experiment. It is, therefore, actually possible to determine at what point in the characteristic curve it is more economical to bank one or more boilers than to carry all at a lower rating.

For instance, assume a station containing 10 boilers, each rated at 600 boiler horsepower. Assume further that all 10 are operated at moderate loads above rating during the day and early evening, and

that later the load on the plant drops to a point which requires 3,000 boiler horsepower for its operation.

This load could be carried in any one of the following ways:

10 boilers working at 50 per cent of rating;
9 boilers working at about 55.5 per cent of rating;
8 boilers working at about 62.5 per cent of rating;
7 boilers working at about 71.5 per cent of rating;
6 boilers working at about 83.5 per cent of rating;
5 boilers working at rating;
4 boilers working at 125 per cent of rating; or
3 boilers working at about 166 per cent of rating.

The operator must determine which one of these various possible arrangements gives the best results. A first tentative decision may be made on the basis of fuel economy and then the effect of other relevant matters may be considered.

CALCULATION OF THERMAL COSTS.

Before any calculations can be made it is necessary to know the fuel cost of banking and the heat value of the coal. Assume, therefore, that it has been found by experiment that it costs 0.2 pound of coal per rated boiler horsepower, per hour, for the character of bank that must be used in the station in question, and that the fuel used has a heat value of 13,500 British thermal units per pound; assume, also, that the lower curve of figure 4 represents the characteristics of the boilers under consideration; then, if a boiler horsepower is assumed to be equivalent of 33,479 British thermal units, it becomes possible to tabulate the thermal costs for all the possible cases, as follows:

1. OPERATION OF 10 BOILERS AT 50 PER CENT OF RATING.

The curve assumed indicates an efficiency of about 57.5 per cent at this loading, therefore

Heat supplied operating boilers per hour $= \frac{3,000 \times 33,479}{0.575} = 174,673,000$

Heat supplied banked boilers per hour $= 0$

Total heat supplied per hour $= 174,673,000$

2. OPERATION OF 9 BOILERS AT 55.5 PER CENT OF RATING.

The curve assumed indicates an efficiency of about 60 per cent of this loading, therefore

Heat supplied operating boilers per hour $= \frac{3,000 \times 33,479}{0.6} = 167,395,000$

Heat supplied 1 banked boiler per hour $= 600 \times 0.2 \times 13,500 = 1,620,000$

Total heat supplied per hour $= 169,015,000$

3. OPERATION OF 8 BOILERS AT 62.5 PER CENT OF RATING.

The curve assumed indicates an efficiency of about 63 per cent at this loading, therefore

Heat supplied operating boilers per hour = $\frac{3,000 \times 33,479}{0.63}$ = 159,423,000

Heat supplied 2 banked boilers per hour = $2 \times 600 \times 0.2 \times 13,500$ = 3,240,000

Total heat supplied per hour = 162,663,000

4. OPERATION OF 7 BOILERS AT 71.5 PER CENT OF RATING.

The curve assumed indicates an efficiency of about 65.2 per cent at this loading, therefore

Heat supplied operating boilers per hour = $\frac{3,000 \times 33,479}{0.652}$ = 154,044,000

Heat supplied 3 banked boilers per hour = $3 \times 600 \times 0.2 \times 13,500$ = 4,860,000

Total heat supplied per hour = 158,904,000

5. OPERATION OF 6 BOILERS AT 83.5 PER CENT OF RATING.

The curve assumed indicates an efficiency of about 68.2 per cent at this loading, therefore

Heat supplied operating boilers per hour = $\frac{3,000 \times 33,479}{0.682}$ = 147,268,000

Heat supplied 4 banked boilers per hour = $4 \times 600 \times 0.2 \times 13,500$ = 6,480,000

Total heat supplied per hour = 153,748,000

6. OPERATION OF 5 BOILERS AT RATING.

The curve assumed indicates an efficiency of about 71 per cent at this loading, therefore

Heat supplied operating boilers per hour = $\frac{3,000 \times 33,479}{0.71}$ = 141,461,000

Heat supplied 5 banked boilers per hour = $5 \times 600 \times 0.2 \times 13,500$ = 8,100,000

Total heat supplied per hour = 149,561,000

7. OPERATION OF 4 BOILERS AT 125 PER CENT OF RATING.

The curve assumed indicates an efficiency of about 72 per cent at this loading, therefore

Heat supplied operating boilers per hour = $\frac{3,000 \times 33,479}{0.72}$ = 139,496,000

Heat supplied 6 banked boilers per hour = $6 \times 600 \times 0.2 \times 13,500$ = 9,720,000

Total heat supplied per hour = 149,216,000

8. OPERATION OF 3 BOILERS AT 166 PER CENT OF RATING.

The curve assumed indicates an efficiency of about 70 per cent at this loading, therefore

Heat supplied operating boilers per hour = $\frac{3,000 \times 33,479}{0.70}$ = 143,481,000

Heat supplied banked boilers per hour = $7 \times 600 \times 0.2 \times 13,500$ = 11,340,000

Total heat supplied per hour = 154,821,000

Obviously methods 6 and 7 are practically equivalent from a thermal viewpoint and are thermally more efficient than any of the other combinations. If no other considerations led to a different result this plant should therefore bank five or six boilers at night.

From the preceding paragraphs it should be apparent that it is possible to base the running of a boiler room on characteristic curves in much the same way as was shown to be possible with the turbine room. However, there is a most essential difference between the two kinds of apparatus. A turbogenerator automatically reproduces a given thermal efficiency at a given load under given conditions of pressure, superheat, and vacuum, but there is nothing automatic about a boiler. The reproduction of results which have been found possible in a boiler room requires the constant attention of a skilled and appreciative operator, one who understands and who has the patience and persistence required to make numerous adjustments at frequent intervals. The greatest savings can generally be effected by so picking and so training the boiler-room crew that the results of daily operation practically duplicate what has been found possible during tests conducted under such conditions as were outlined in an earlier paragraph.

Experience has shown that one of the best methods of keeping a boiler-room crew up to the proper working level is to determine average boiler efficiency for each week or each month of operation and to acquaint them with the results. Such efficiencies are easily calculated with a reasonable degree of accuracy if the coal is weighed and the feed water is metered. Equipment for such purposes should form an integral part of every modern station.

OPERATION OF AUXILIARIES.

In steam power-plants power is required for running a number of auxiliaries, such as coal-handling apparatus, stokers, boiler feed pumps, circulating pumps, etc. The methods used for obtaining this power are determined as a compromise between reliability, convenience, thermal economy, and financial economy.

Before the various methods available are considered it is desirable to call attention to several impressions that are widely held, although they are only partly true. Thus it is often assumed that steam-driven auxiliaries are thermally the most economical because the steam exhausted from auxiliary drives can be used in heating the feed water. Again, others hold that electric drive is thermally the most economical arrangement because the power used by the auxiliaries is generated in the main units of the station and these are unquestionably far more efficient than the scattered small units that would be used for direct steam drive. Similarly, there are several other equally comprehen-

sive statements of supposed facts with regard to auxiliary drive that are just as mutually contradictory.

As a matter of fact, no broad statements of the type indicated above can be true in general. Each method available is adapted to the solution of certain problems and it generally results in practice that a compromise must be made between several methods to obtain the best results. Certain fundamentals can be laid down, however, and may be used as guiding principles for making a choice between possible methods in planning a new installation, or as indications of the best methods of operating stations already constructed. Often a study of operating conditions in the light of these fundamentals will show that it would pay handsomely to alter the auxiliary machinery in existing plants.

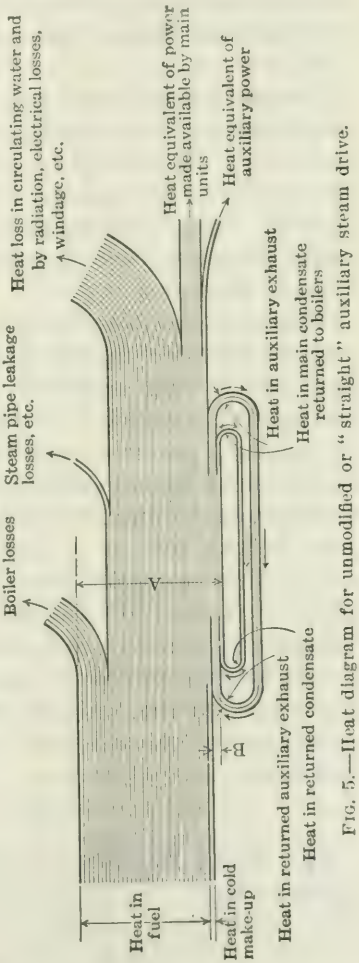
Heat loss in circulating water and by radiation, electrical losses, windage, etc.
Heat equivalent of power made available by main units
Heat equivalent of auxiliary power
Steam pipe leakage losses, etc.
Heat in auxiliary exhaust
Heat in main condensate returned to boilers
Boiler losses
A
B
Heat in returned auxiliary exhaust
Heat in returned condensate
Heat in fuel
Heat in cold make-up

FIG. 5.—Heat diagram for unmodified or "straight" auxiliary steam drive.

"STRAIGHT" STEAM DRIVE.

What may be designated as the unmodified or "straight" steam method of auxiliary drive is illustrated diagrammatically in figure 5. The width of the stream as shown by the dimension A represents the total amount of heat supplied the boiler. Of this, a part is lost at the boiler, and a part is lost in steam lines and traps, and a part enters the main unit. The remainder finally arrives at the auxiliary machinery, where it is split into three parts: (*a*) The auxiliary power sought; (*b*) losses by radiation, friction, windage, etc.; and (*c*) heat in the exhaust steam. The heat in the exhaust steam is returned to the boiler by being absorbed in the feed water, and thus it forms part of the width of the stream as indicated by A. If it were not returned in this way the part of the stream representing heat in fuel would have to be wider by the amount indicated by B in the figure.

The heat in the exhausted auxiliary steam really travels around in a short-circuited path as indicated. It joins the stream going to the auxiliaries, thus increasing its width, and continuously comes back and starts again.

As a rough approximation, auxiliary power obtained in this way may be said to cost about 4,500 British thermal units per kilowatt-hour of auxiliary power. This figure is obtained as follows: One kilowatt-hour is equivalent to 3,412 British thermal units, and the net cost of obtaining this with the method shown in figure 5 is 3,412 British thermal units plus some boiler losses, steam-pipe losses, and radiation, friction, and windage losses chargeable to 1 kilowatt-hour of auxiliary power. The total is ordinarily not much different from 4,500 British thermal units in fuel per kilowatt-hour of auxiliary power.

It must be noted that this figure holds only for conditions under which the steam exhausted from auxiliaries can be absorbed by the feed water. If a condition arises in which auxiliary exhaust must be discharged to the atmosphere, the auxiliary power must be charged with the heat in its exhaust steam, and a kilowatt-hour generated under these conditions involves an expenditure of 50,000 to 200,000 British thermal units.

It is now possible to demonstrate the fallacy of a common statement to the effect that the economy of the steam engines or turbines used for driving auxiliary apparatus makes no difference in the economy of the plant. In plants of modern proportions the amount of auxiliary power required per kilowatt capacity of main units is such that at light and moderate loads much more auxiliary steam would be made than could be absorbed by the feed water. It therefore follows that if a straight steam-driven auxiliary system is adopted, the greatest thermal efficiency will result when the most efficient engines or turbines are used for driving auxiliaries. Whether the use of such efficient, and therefore costly, auxiliary drives would result in the greatest economic efficiency would depend entirely upon the costs of fuel and capital. Ordinarily, the fuel saving would be such as to balance a very considerable increase in first cost.

"STRAIGHT" ELECTRICAL DRIVE.

The direct opposite of what has been called the unmodified steam method will be designated as the unmodified or "straight" electrical method. In this method all auxiliaries are motor driven and the power is taken from the main units served or from the station bus. Turbine plants of modern design should produce a kilowatt-hour with an expenditure of 19,000 to 25,000 British thermal units in coal. Therefore every kilowatt-hour taken for the driving of auxiliaries represents a thermal cost of the order of 23,000 to 30,000 thermal units, after allowing for the various efficiencies involved.

The unmodified electrical method possesses many advantages, such as reliability, simplicity, flexibility, cleanliness, and such, but it is

unquestionably thermally expensive as compared to the other method under conditions that permit total absorption of the exhaust. On the other hand, the thermal cost of auxiliary power generated without reclamation of the heat in the exhaust is two to eight times as large as that characteristic of the "straight" electrical method.

In some of the stations installed a decade or more ago the auxiliary problem was solved by the use of both steam and electrically driven units. This may be designated as the dual system. A sufficient number of steam-driven auxiliaries were operated to give that quantity of exhaust steam which could be absorbed by the station feed, and all of the remaining auxiliary power was obtained electrically with energy taken from the main units. Many systems of this sort are in successful operation.

Obviously, the correct apportionment between the two different methods in the dual system varies with the station load. With high loads and large quantities of feed water the greater part of the auxiliary power is obtained with steam, and with light loads a large part is obtained electrically. Consequently the thermal cost of auxiliary power varies with the load. At high loads this cost tends to approach the 4,500 to 5,000 thermal units characteristic of steam drive with utilization of exhaust, and at light loads it tends to approach a figure nearer 20,000 to 30,000 thermal units.

Following the development of the dual system came many schemes of various sorts for still further reducing the cost of auxiliary power and for increasing operating convenience.

OTHER SYSTEMS FOR DRIVING AUXILIARIES.

One such system drives the auxiliaries electrically with power taken from the main units and heats the feed water with steam drawn from the medium or low pressure stages of the main turbines. This system is capable of many variations but in all of its forms it amounts, thermally, to the generation of auxiliary power with steam up to the ability of the feed water to absorb the exhaust, and the use of electrical energy generated in the main unit for auxiliary power in excess of this quantity. However, as the main units are the source of the steam-generated part of the auxiliary power, the steam drives of the auxiliaries are very economical and a larger proportion of auxiliaries can be driven in this way than could be done if many small steam-driven units were used. Moreover, all advantages of "straight" electrical drive are obtained, and the operation of the system is delightfully simple. The different demands made by the feed water at different loads can be met by merely controlling the steam bled from lower stages of the turbine. By placing the bleeder valves under thermostatic control the operation can be made entirely automatic.

The thermal cost of auxiliary power with this arrangement is similar to that of the dual system, but with normal load curves it should be lower through the greater thermal efficiency of the apparatus used for furnishing the "steam-driven" part of the auxiliary power. Because of the greater thermal efficiency, a smaller number of pounds of "exhaust" steam will be made per unit of auxiliary power generated and a greater part of the total can therefore be generated in this way without exceeding the ability of the feed water to absorb "exhaust" steam.

Another system that has been used extensively is almost the opposite of that last described. The auxiliary apparatus is driven by numerous small steam-operated units. As much as possible of the steam exhausted by these units is absorbed in the feed water and the remainder is fed to the low-pressure stages of the main turbines. By the use of an automatic valve on the auxiliary exhaust the apportionment of exhaust steam can be made automatic in operation; in this respect the system behaves similarly to that last described.

It is obvious that whatever part of the auxiliary power is derived from steam that is later absorbed by the feed water is obtained at the thermal cost characteristic of a "straight" steam system. As the steam drives of the auxiliaries are bound to be of small capacity they are commonly characterized by low thermal efficiency and therefore use a relatively large quantity of steam per kilowatt-hour developed. In this respect the system is less economical than that previously described, because with absorption of exhaust steam by the feed water a smaller part of the total auxiliary demand can be supplied.

A consideration of the thermal cost of that part of the auxiliary power obtained from steam exhausted into the main units is more complicated. In general, it can be assumed that about half of the energy made available by steam expanding from boiler to condenser pressure in modern turbines is developed during the expansion from boiler pressure to atmospheric pressure. The other half is produced by the expansion from atmospheric pressure to the back pressure maintained at the turbine exhaust. Therefore, every pound of auxiliary steam exhausted at about atmospheric pressure into the lower stages of the main turbines may be said to make available at the main generators about the same amount of energy as would be made available by half a pound of steam entering at the high-pressure end of the main turbines and expanding to condenser pressure.

The thermal cost of auxiliary power obtained from steam exhausted into the main units may obviously be obtained by subtracting from the heat supplied the auxiliaries per kilowatt of auxiliary power a credit based on the amount of heat that has been made unnecessary for the supply of the main units. Thus, if the auxiliaries

use, say, 50 pounds of high-pressure steam per kilowatt-hour of auxiliary power generated, this steam exhausted into the main units will take the place of 25 pounds of main steam. The net cost of 1 kilowatt-hour of auxiliary power would then be the heat above feed temperature in 25 pounds of high-pressure steam.

With feed at 210° F. and modern pressures and superheats; the quantity of heat thus chargeable per pound would be in the neighborhood of 1,100 British thermal units. In the assumed case the auxiliary power would therefore cost about $25 \times 1{,}100 = 27{,}500$ British thermal units per kilowatt-hour. With more efficient auxiliary drives this figure would be lowered and with less efficient auxiliary drives it would be raised.

The values assumed are somewhat lower than would be expected in the average practice, and it is obvious that, in general, this method of obtaining auxiliary power may be expected to be slightly more costly than that last described.

CONCLUSIONS.

A careful consideration of all the methods described thus far warrants these conclusions:

1. The smallest thermal cost for auxiliary power is incurred when generated by steam with complete absorption of the exhaust in the feed water.

2. The greatest possible part of all of the auxiliary energy required can be obtained in this way when the steam drives used with the auxiliaries have the greatest possible thermal efficiency.

3. Auxiliary power in excess of that obtainable with exhaust steam absorption can be procured from the main generators in electrical form at a lower thermal cost than in any other way.

To these conclusions may now be added two commonly known facts:

1. The increasing price of fuel is leading more central stations to the use of economizers.

2. The greatest saving is effected with economizers when the water supplied them is at the lowest permissible temperature. This value is probably somewhere between 120° and 150° F., depending upon the types of economizer used and the care with which it is operated.

At first glance some of the items listed in the two groups seem contradictory when viewed with regard to thermal efficiency. Thus the use of feed water at 140° F., say, must reduce the quantity of auxiliary steam that can be used for feed-water heating and must therefore increase the average cost of auxiliary power. However, when all of the listed items are taken into account they point toward another solution of the auxiliary power problem which has been found both thermally efficient and very practical.

It is obvious that fairly high thermal economy is more readily obtainable with one turbine of moderate size than with a great number of small turbines aggregating the same capacity. It is also obvious that, if the turbine is properly designed, the quantity of steam required by it at any given load will decrease as the back pressure is decreased.

With the system referred to, all, or practically all, of the auxiliaries are electrically driven, but as much as possible of the electric power used is taken from an auxiliary, steam-driven generator of fairly large size. The remainder is taken from the main unit. Steam exhausted by the auxiliary turbogenerator unit is passed into a feed-water heater as in the "straight" steam method, but the feed-water heater is arranged so that it can be operated at a pressure below atmospheric if desired.

The feed-water heater therefore may serve as a condenser so far as the auxiliary turbine is concerned and the water rate of the auxiliary turbine may thus be lowered still further, with reference to the aggregate water rate of a number of small steam units.

This system is thermally the exact counterpart of the dual method previously described. It is, however, thermally superior in that the greater efficiency of the large auxiliary unit makes possible the absorption of the steam made from the generation of a larger part of the total auxiliary power. It is also superior in that a control of the auxiliary steam made is available, for, with any given load, the quantity of steam demanded depends entirely on the back pressure.

These points of superiority can be utilized in two ways. The auxiliary unit may be operated with atmospheric-exhaust pressure, when the advantage gained is merely that of being able to carry a greater part of the total auxiliary load on the steam-driven auxiliary apparatus. Or the auxiliary unit may be operated with low back-pressures, so low as to give feed-water temperatures as low as 140° F.; in this case economizers can be used to full advantage and a greater part of the auxiliary power can be carried on steam-driven machinery than would be permissible with a number of small steam-driven units exhausting at a corresponding temperature.

Thermally, the cost of auxiliary power is approximately 4,500 to 5,000 thermal units for all that power obtained from the auxiliary generator, and all power in excess of this is obtained from the main generator at a thermal cost of 23,000 to 30,000 heat units.

From this hasty survey of the methods of obtaining auxiliary power, and the possible costs of auxiliary power it should be evident that large errors can be made in the original design. It is also possible to make costly errors in operation. To illustrate: Assume that in a certain station, operating at normal day load, the total amount of power used for auxiliaries and for house service of all kinds

amounts to 10 per cent of that sent out. This is a figure that should not be exceeded but is often equaled. Under these conditions, for every 1,000 kilowatt-hours sent out of the station it is necessary to produce 100 kilowatt-hours for the operation of auxiliaries, station lights, etc.

If it be assumed that the main units are capable of producing a kilowatt-hour for 20,000 thermal units in fuel fired it is possible to show the decidedly weighty influence of the thermal cost of auxiliary power.

It has been shown that auxiliary power may cost anywhere between about 4,500 and 200,000 thermal units, depending on the method used. At 4,500, the auxiliary power would cost about $4,500 \times 100 = 450,000$ thermal units in the assumed case. This would be equal to about 2.25 per cent of $20,000 \times 1,000 = 20,000,000$ thermal units required by the main units. On the other hand, if the auxiliary power costs 200,000 units per kilowatt-hour the total would be $200,000 \times 100 = 20,000,000$, or exactly the same as that required by the main units.

With all auxiliary power taken electrically from efficient main units the cost would probably be at least 23,000 units per kilowatt-hour. This would give a cost in the assumed case equal to $23,000 \times 100 = 2,300,000$, or about 11.5 per cent of that required by the main units for the generation of power leaving the station.

These figures should serve to show the tremendous importance of the proper operation of auxiliary machinery. It does little good to determine the water rates of the main units and operate for a minimum station water rate of these alone, as suggested in the first part of this paper, if the thermal savings effected are equaled or more than equaled by losses through wrong operation of auxiliary systems. Further, the tendency to regard these auxiliary systems as necessary nuisances of no great thermal significance is very apt to lead toward such a state.

In conclusion, as much as possible of the auxiliary power should be obtained in such a way that resultant exhaust steam can be absorbed by the feed water. The remainder of the necessary auxiliary power should be obtained in electrical form from the most efficient units available. The observance of these two rules leads to maximum thermal economy.

The exact method adopted for achieving these desirable results must depend upon the layout and the characteristics of the particular station. No one system is preeminently superior under all conditions. However, it should be noted that the possibilities of a system often can be greatly increased by making comparatively insignificant changes along lines suggested in the preceding discussion.

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